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An improved method for tool point dynamics analysis using a bi-distributed joint interface model

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ABSTRACT

The existing tool point dynamics analysis methods may lead to inaccurate predictions of frequency response function (FRF) in some cases since they ignored the collet geometry. This paper presents an improved method to better predict the contributions of collet geometry to FRF of cutting tool by introducing the effect of collet. The spindle–holder–collet–tool assembly is modeled as two distributed joint interfaces, i.e., collet–holder and collet–tool joint interfaces, rather than the existing single holder–tool joint interface without the effect of collet. Dynamics of the tool and the collet are analyzed using Euler–Bernoulli beam theory, and the tool–collet and holder–collet joint interfaces are separately treated as two distributed zero-thickness damped-elastic layers. The contact stiffness and damping properties of both joint interfaces are identified by minimizing the discrepancy between the measured and predicted tool point FRFs. The tool–collet assembly is supposed to rest on the resilient support provided by the spindle–holder assembly, whose dynamical property is analytically calculated by the receptance coupling substructure analysis (RCSA) method. Wider prediction capacity of the proposed method has been experimentally verified via the comparison with traditional method.

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1. Introduction

Chatter vibration in machining process is recognized as the basic factor to cause many negative effects such as poor surface finish, unacceptable inaccuracy, reduced material removal rate, disproportionate tool wear, machine tool damage and so on [1]. To ensure a stable cutting, many strategies such as prediction of chatter stability in advance, installation of extra energy absorption devices and disruption of regenerative chatter have then been summarized in [2]. Among these, the most cost- and time-saving method is to select chatter-free cutting parameters from the stability lobe diagram (SLD) based on the frequency response function (FRF) of tool point [3–12]. For instance, Altintas and Budak [5] presented a frequency-domain method for the analytical prediction of SLD in a milling process. Insperger et al. [9] proposed a semi-discretization method to construct SLD for milling at small radial immersions. Totis et al. [10] developed a fast algorithm for chatter prediction in milling with spindle speed variation based on the Chebyshev Collocation Method. Kecik et al. [11] developed a nonlinear model to predict chatter vibration in the high-speed milling processes considering both regeneration and frictional chatter. Rusinek et al. [12] utilized the recurrence plot technique and the Hilbert–Huang transformation to identify the chatter vibration in milling of titanium superalloy.

Generally, the FRF of the tool point is case-dependent and obtained by impact testing. In other words, the test should be repeated once the sizes of holder and tool change. Moreover, measurements of the FRFs are almost impractical for micro-machining tool. Hence, developments of general computing methods become a vital alternative for the determination of FRF of tool point. This motivates the development of tool point dynamics analysis methods in the current work.

Historically, three important development stages existed on the calculation of tool point FRF.

- Lumped non-rigid joint model involving two substructures: Schmitz and Donaldson [14] proposed the first-generation of receptance coupling substructure analysis (RCSA) method that treated the spindle–holder–tool assembly as two substructures consisting of spindle–holder substructure and the overhung part of tool substructure, as shown in Fig. 1(a). In this method, the joint of the two substructures was modeled as translational spring and damper with the dynamical properties determined by fitting approach. Efforts have been made later to improve the accuracy of the method [15–19]. For example, Park et al. [15] analytically calculated the rotational FRF considering both lateral and rotational dynamic responses of the spindle–holder

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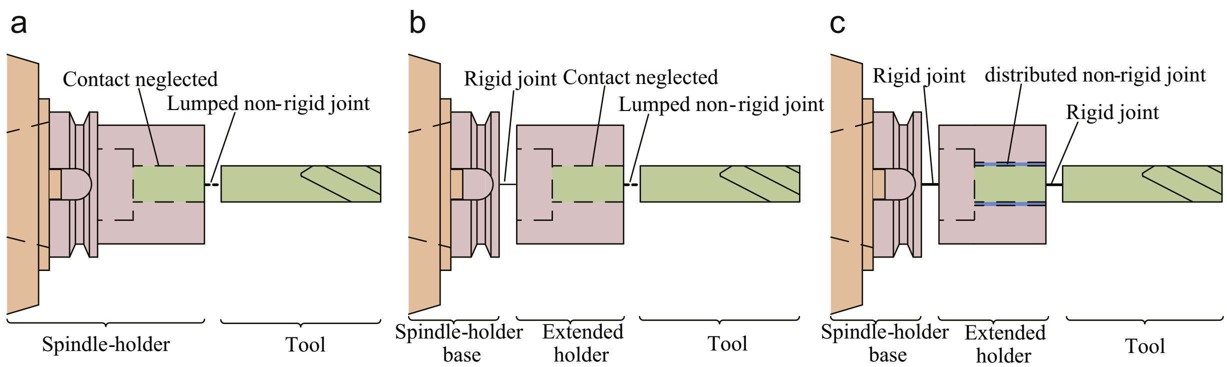

Fig. 1. Different partition strategies for a spindle-holder-tool system reported in [13]: (a) Lumped non-rigid joint model involving two substructures; (b) Lumped non-rigid joint model involving two substructures and (c) Distributed non-rigid joint model.

substructure, while Albertelli et al. [18] used a finite difference method to consider the effect of both lateral and rotational dynamic responses of the spindle–holder substructure. Kivanc and Budak [16] considered the effect of the complex geometry of flutes using stepped Euler–Bernoulli beam. Park and Chae [17] utilized Euler–Bernoulli beam to calculate the tool point receptances of modular cutting tools including the dynamic effect of fastener joint, which was identified by means of analytical and experimental analyses. Mancisidor et al. [19] predicted the tool point dynamics using fixed boundaries approach and pointed out that the cut-off frequency problem could be overcome in the receptance coupling procedure.

- Lumped non-rigid joint model involving three substructures: Schmitz and Duncan [20] proposed the second-generation of RCSA method. The basic idea is to consider the spindle–holder–tool assembly as three substructures: the spindle–holder base, the extended holder and the overhung part of tool, as shown in Fig. 1(b). In this method, the receptances of spindle–holder base were determined by impact testing since theoretical modeling of the spindle-machine portion was no longer achievable. Meanwhile, the spindle–holder base and extended holder were rigidly coupled, while the joint between extended holder and the overhung part of tool was still modeled as a combination of spring and damper. Receptances of the extended holder and the overhung part of tool were calculated using an analytical method. For instance, Ertürk et al. [21,22] adopted Timoshenko beam theory to analytically model the spindle–holder–tool dynamics, and proved that the accuracy of the model at high frequency could be improved in such a way. Ozsahin et al. [23] investigated an identification method of dynamical contact parameters using Ertürk's model. Filiz et al. [24] utilized stepped beam to model the complete holder–tool substructure so that the actual tapered geometry of a shrink fit holder was accurately described. Bediz et al. [25] adopted a spectral–Tchebyshev technique to take into account the effect of actual fluted geometry on three-dimensional dynamic behavior of milling tools.
  However, the actual contact effect between the tool and holder interface was ignored in all the aforementioned work, and a lumped joint was assumed between the overhung part of tool and inserted part of tool. In fact, dynamical interactions always exist along the whole tool–holder joint interface.
- Distributed non-rigid joint model: This is the third generation of RCSA method that modeled the tool–holder joint interface as distributed joint, as illustrated in Fig. 1(c). Movahhedy and Gerami [26] used linear joint elements to equivalently simulate the rotational stiffness of the joint, which was further identified by genetic algorithm. Schmitz et al. [13] developed multiple connection models for the tool–holder interface, and employed finite element method to determine the position-dependent stiffness and equivalent viscous damping values for a thermal shrink fit holder. Ahmadi and Ahmadian [27] and Ahmadian and Nourmohammadi [28] combined the measured FRFs of spindle–holder and analytical models of the tool via a distributed damped-elastic tool–holder interface. In their methods, experiments had to be repeated for holder size changes since dynamics of each specific spindle–holder was obtained by impact testing. Yang et al. [29] presented a generalized method to predict the tool point bending, torsional and axial receptances of all kinds of rotating tools by using three-dimensional Timoshenko beam theory.
  Notice that in the work stated above [13,26–29], only one damped-elastic layer was used to model the holder–collet–tool connection system, and the physical attributes of collet were ignored in the model.
  This paper presents a systematic method to predict the FRFs of tool point by considering the effect of collet. The tool–collet and holder–collet joint interfaces are considered as two distributed layers with varying stiffness. The tool is assumed to partly rest on the collet via a distributed damped-elastic tool–collet interface while the collet is assumed to rest on the resilient support provided by the spindle–holder assembly via a distributed damped-elastic holder–collet interface. Stiffness and damping properties of both joint interfaces are identified by minimizing the discrepancy between the measured and predicted FRFs of tool point. More importantly, a computing procedure is proposed to eliminate repeated impact tests in obtaining the dynamics of the spindle–holder assembly of different sizes. A total routine is given to compute the FRF of tool point. Besides, the influence of the collet on the prediction accuracy is investigated by comparing the proposed method with the existing method. The method addressed in this paper is experimentally verified for different spindle–holder–tool assemblies.

2. Tool point dynamics analysis method

Mathematically, the FRF of the tool point is defined as:

$$H(\omega)=\frac{Xe^{i\omega t}}{Fe^{i\omega t}} \tag{1}$$

where $Xe^{i\omega t}$ is the dynamic displacement of the tool point under harmonic force $Fe^{i\omega t}$. t and ω denote time and angular frequency, respectively. It can be seen that the key issue related to the FRF is how to obtain the solution of $Xe^{i\omega t}$ under $Fe^{i\omega t}$. Based on Euler–Bernoulli beam theory, a new procedure to solve $Xe^{i\omega t}$ is presented in this section.

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